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Effect of different flow regime on the static and dynamic performance parameter of hydrodynamic bearing

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Abstract

The hydrodynamic journal bearing system has found wide spread application in high speed rotating machine such as compressors, gas turbines, water turbines, steam turbines, alternators etc. As rotor generally operates at high speed, the lubricant flow in the clearance space of journal bearing does not remain laminar and for accelerated/ decelerated journals the threshold speed of instability is crossed from both sides. In this paper the numerical method has been used to compute the static and dynamic performance parameter. The analysis is carried out for the case of short bearing approximation aspect ratio (L/D≤0.5) under different fluid flow regime i.e. laminar, transition and turbulent flow condition assuming the perfectly rigid journal and bearing.

Keywords: Fluid film journal bearing, Laminar flow, Turbulent flow, static & dynamic performance.

Nomenclature

a	Tangential acceleration
C	Diameter of clearance
C_{ij}	Fluid film damping coefficient
D	Diameter of journal bearing
h	Minimum fluid film thickness
L	Length of bearing
R	Radius of journal
t	Time
t_r	Reference time
U	Instantaneous peripheral speed
ε	Eccentricity ratio
K_x	Turbulence coefficient in x axis
K_y	Turbulence coefficient in y axis
K_{ij}	Fluid film stiffness coefficient
X_j, X_j	Instantaneous Journal Centre Co – ordinate
$\bar{p}$	Dimensionless pressure
W	Load bearing capacity
Φ	Attitude angle
e	Eccentricity ratio
μ	Dynamic viscosity
μ_r	Reference viscosity
α	Angle measured from positive axis
α_1, α_2	Angle measured from x axis to start and end of positive pressure zone

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1. Introduction

Hydrodynamic journal bearing is a bearing operating with hydrodynamic lubrication, in which the bearing surfaces are separated from the journal surface by the lubricant film, generated by the journal rotation. The operation of a hydrodynamic journal bearing depends upon the shearing of a film of lubricant in the clearance space between the bearing and the journal load supporting pressure is generated within the film by continuous rotation of the journal. Hydrodynamic bearing is used, when following conditions have to be met.

1.) The mating surface must not be parallel.
2.) A sufficient relative velocity must exist.

The development of hydrodynamic pressure in the clearance space is governed by Reynolds equation, that can be solved in closed form, considering boundary conditions, by either of two approximation-infinitely long bearing approximation (valid for aspect ratio, L/D>4) or short bearing approximation (valid for L/D ≤ 0.25). For finite bearings (0.25 < L/D ≤ 4) Reynolds equation is solved numerically. For dynamically loaded bearing, short bearing approximation is generally used (for L/D≤ 1) to save computing time. Reynolds equation is modified for super laminar flow inside the clearance space, by including turbulence coefficients, K_x and K_y, from linearised turbulence theories. In addition, transition from laminar to vortex flow and to turbulence is evaluated by using concept of mean and local Reynolds number. After getting pressure distribution along the mean circumference of clearance space, load carrying capacity, which is the maximum load, a given bearing is able to support, without film failure, can be found, for static equilibrium of journal. At this position attitude angle, which is the angle between line of action of vertical load and line joining the bearing centre and journal centre, is evaluated. From this, circumferential position of the minimum film thickness is determined.

For dynamic case, fluid film is idealised as four springs and four dampers and thus fluid film stiffness and damping coefficients are evaluated. Stability analysis can be done by applying Routh's criteria on the equation of motion for free vibration and critical mass, threshold speed at the onset of instability can be determined.

Using short bearing approximation(SBA) for derivation of the Reynolds equation with usual assumption, Kirk.et.al [1], get the rapid solution of the fluid film bearing forces and found the assumption to be very good for L/D ratios of 0.5 or less. Barrett et al. [2] introduced a finite length correction factor, which modifies the nonlinear forces from short bearing theories for finite length bearings. Rezvani et al. [3] modified the short bearing approximation and presented this Modified Short Bearing Approximation (MSBA) by approximating axial pressure profile parabolic in terms of P_s and P_c which are functions for eccentricity, angular position of any point from line of centre along bearing surface and non dimensionlized vertical co-ordinate for any given L/D ratio. A review of existing turbulent lubrication theories and their application to fluid film bearing design which was given by Taylor et al. [4]. Frene et al. [5] showed that the actual transition region may not be sufficiently wide to include many of the practical application. A concept was given for evaluation of transition from laminar to vortex flow and to turbulence using mean and local Reynolds number. After getting the flow type, with help of corresponding effective Reynolds number, parameters $\overline{K_x}$ and $\overline{K_y}$ can be evaluated. Thus, in the full range of operating condition, Reynolds equation can be solved.

To analyze the dynamic characteristics of turbulent journal bearing, Hashimoto et al. [6] had considered short bearing theory and turbulent lubrication theory of Ng and Pan [7] and the results were compared to the finite bearing theory. Variation in the whirl onset velocity with Sommerfeld number under the laminar and turbulent lubrication condition showed that the whirl onset velocity for relatively low Sommerfeld numbers becomes smaller with increases in Reynolds number. Capone et.al [8] determined the stiffness and damping coefficients and the stability limit curves for various Reynolds number for a given value of clearance ratio (C/R) ratio of a journal bearing in a non- laminar lubrication regime. Analysis of turbulent journal bearing by [6, 8] had been done by considering the flow to be either laminar or fully developed turbulent. Capone et al. [9] examined flow between laminar and fully developed turbulent. During starting and stopping of hydrodynamic journal bearings the wear between shafts and bearing surfaces is of prime importance. In the experimental work, Mokhtar et al. [10] found that under typical start up conditions, hydrodynamic forces are generated very rapidly and affect the journal behaviour. If initial rolling occurs at all, its extent is only small and maximum attitude angle is much less than the metallic friction angle. Malik et al. [11] observed that theoretical prediction of stopping behaviour is the same as the experimental and is not difficult to derive but the theoretical prediction of starting behaviour is not directly derivable. During starting the development of full hydrodynamic film precedes three stages, namely, a small amount of riding of the journal over the bearing, the boundary lubrication and the quasi-hydrodynamic lubrication.

To the best of the author's knowledge no study has been reported on the analysis of static and dynamic performance of journal bearing simultaneously for all three fluid flow regimes i.e. laminar, transition and turbulent. Therefore, the objective of this study was to bridge this gap and to study the effect of different flow regimes on static and dynamic performance parameter of hydrodynamic bearing. The bearing performance has been compared on the basis of the same L/D and C/R. The results presented in this study are expected to be very useful for bearing designers.

2. Flow field equation

The Reynolds equation which governs the flow of lubricating oil in the clearance space of a hydrodynamic journal bearing using linearized turbulent theory

$$\frac{\partial}{\partial x}\left[\frac{h^3}{\mu K_x}\frac{\partial p}{\partial x}\right]+\frac{\partial}{\partial y}\left[\frac{h^3}{\mu K_y}\frac{\partial p}{\partial y}\right]=\frac{1}{2}U\frac{\partial h}{\partial x}+\frac{\partial h}{\partial t} \tag{1}$$

Where, $U=[\Omega y + a(t-t_r)]R$

After non dimensionlization it reduces to

$$\frac{\partial}{\partial \alpha}\left[\frac{\bar{h}^3}{\bar{\mu}\bar{K}_\alpha}\frac{\partial \bar{p}}{\partial \alpha}\right]+\frac{\partial}{\partial \beta}\left[\frac{\bar{h}^3}{\bar{\mu}\bar{K}_\beta}\frac{\partial \bar{p}}{\partial y}\right]=\frac{1}{2}\bar{\Omega}\frac{\partial h}{\partial \alpha}+\frac{\partial h}{\partial \bar{t}} \tag{2}$$

Where $\bar{h}=1.0-\bar{X}_j\cos\alpha-\bar{Z}_j\sin\alpha$, $\bar{X}_j=\frac{x_j}{c}, \bar{Z}_j=\frac{z_j}{c}, \bar{h}=\frac{h}{c}$ and X_j and Z_j are the coordinates of journal centre, So

$$\frac{\partial \bar{h}}{\partial \alpha}=\bar{X}_j\sin\alpha-\bar{Z}_j\cos\alpha \tag{3}$$

$$\frac{\partial \bar{h}}{\partial t}=-\bar{X}_j\cos\alpha-\bar{Z}_j\sin\alpha \tag{4}$$

So put the value of eq.3 eq.4 in eq.2 can be written as follows

$$\frac{\partial}{\partial \alpha}\left[\frac{\bar{h}^3}{\mu K_\alpha}\frac{\partial \bar{P}}{\partial \alpha}\right]+\frac{\partial}{\partial \beta}\left[\frac{\bar{h}^3}{\mu K_\beta}\frac{\partial \bar{P}}{\partial \beta}\right]=\frac{1}{2}\bar{\Omega}\left(\bar{X}_j\sin\alpha-\bar{Z}_j\cos\alpha\right)-\bar{X}_j\cos\alpha-\bar{Z}_j\sin\alpha \tag{5}$$

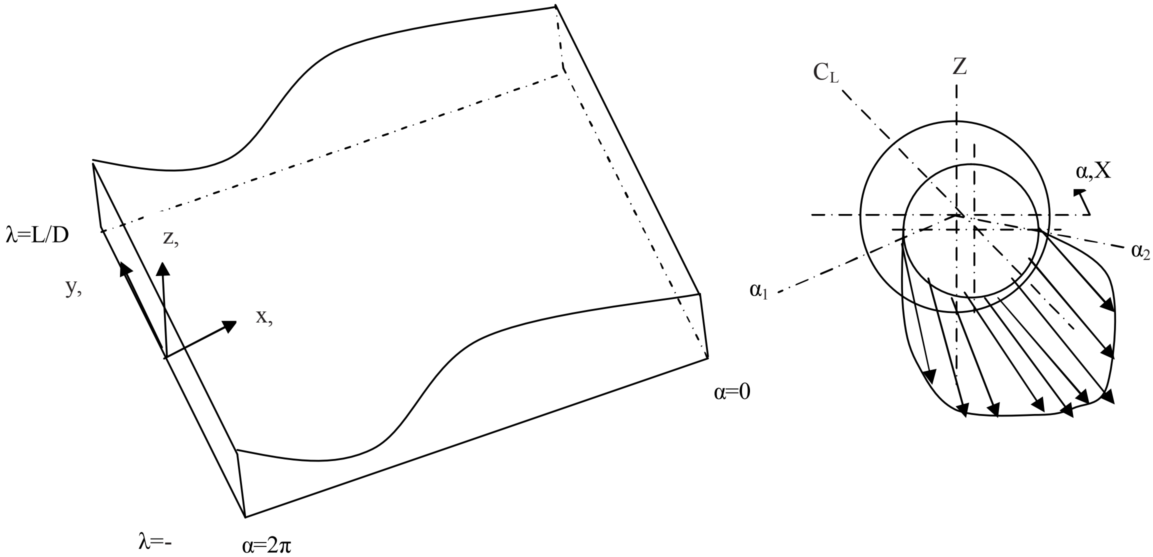

Fig.1: Development of fluid film between bearing and journal surface

The value of K_α and K_β i.e. turbulence coefficient are obtained using following relation [Tayler et.al.(1974)]

$K_\alpha = 12.0 + 0.0136\, Re_e^{0.9}$

$K_\beta = 12.0 + 0.0043\, Re_e^{0.9}$

Where Re_e the effective Reynolds number is given by

$$Re_e=\frac{\rho U h}{\mu}=Re_{ge}\left[\frac{\bar{\Omega}_r+\bar{a}(\bar{t}-\bar{t}_r)}{\bar{\mu}}\right]\bar{h} \tag{6}$$

The relation have been extended to include both laminar and transition regimes as follows

(i) Laminar flow $Re_e = 0$

(ii) Transition flow $Re_e = (Re_m / Re_c - 1).(\rho Uh/\mu)$
(iii) Fully developed turbulent flow $Re_e = \rho Uh/\mu$
Where Re_m= local Reynolds number, Re_{ge} = global or average Reynolds number
Re_c =critical Reynolds number and $Re_m=2\rho q/\mu$; where ρ= specific weight, q= total flow and μ=viscosity of fluid.
With the help of above different Reynolds number the three flow regimes are defined as [5]
(i) Laminar flow condition is $Re_m \le Re_c$
(ii) Transition flow condition is $Re_c \le Re_m \le 2\, Re_c$
(iii) Fully developed turbulent flow condition is $Re_m \ge 2\, Re_c$

2.1. Boundary Conditions

The boundary condition used for the solution of lubricant flow field are described as
i) $\bar{p} = 0\ at\ \alpha = \alpha_1$
ii) $\bar{p} = 0\ at\ \alpha = \alpha_2$
Where α_1, α_2 are the angles measured from positive X axis to the start and end of positive pressure zone

2.2. Performance Characteristics

The bearing performance characteristic parameters of a hydrodynamic journal bearing are obtained by using following expression as reported in References. [5, 6, 8]

Attitude angle (Φ) is defined as the angle between the line of action of the external load 'W' and the line joining the two centres. It is determined by the equilibrium position of journal.

$$(\Phi) = \tan^{-1}[\text{abs}\ \{X_j / Z_j\}] \tag{7}$$

The fluid film stiffness coefficients are defined in matrix form as

$$\begin{bmatrix} \overline{K_{xx}} & \overline{K_{xz}} \\ \overline{K_{zx}} & \overline{K_{zz}} \end{bmatrix} = -\begin{bmatrix} \dfrac{\partial \overline{F_x}}{\partial \overline{x_j}} & \dfrac{\partial \overline{F_x}}{\partial \overline{z_j}} \\ \dfrac{\partial \overline{F_z}}{\partial \overline{x_j}} & \dfrac{\partial \overline{F_z}}{\partial \overline{z_j}} \end{bmatrix} \tag{8}$$

The fluid film damping coefficients are defined in matrix form as

$$\begin{bmatrix} \overline{C_{xx}} & \overline{C_{xz}} \\ \overline{C_{zx}} & \overline{C_{zz}} \end{bmatrix} = -\begin{bmatrix} \dfrac{\partial \overline{F_x}}{\partial \dot{\overline{x}}_j} & \dfrac{\partial \overline{F_x}}{\partial \dot{\overline{z}}_j} \\ \dfrac{\partial \overline{F_z}}{\partial \dot{\overline{x}}_j} & \dfrac{\partial \overline{F_z}}{\partial \dot{\overline{z}}_j} \end{bmatrix} \tag{9}$$

Where F_x and F_z are the film force components at the equilibrium position, which are expressed as follows

$$F_x = -\int_{-L/2}^{L/2} \int_{\alpha_1 R}^{\alpha_2 R} p \cos\alpha\ dx.dz \tag{10}$$

$$F_z = -\int_{-L/2}^{L/2} \int_{\alpha_1 R}^{\alpha_2 R} p \sin\alpha\ dx.dz \tag{11}$$

In non-dimensional form

$$\overline{F_x} = \frac{F_x C^2}{\mu_r \omega_r R^4} = \int_{-\lambda}^{\lambda} \bar{p} cos\alpha\ d\alpha.d\beta \tag{12}$$

$$\overline{F_z} = \frac{F_z C^2}{\mu_r \omega_r R^4} = \int_{-\lambda}^{\lambda} \bar{p} sin\alpha\ d\alpha.d\beta \tag{13}$$

3. Solution Procedure

Reynolds equation which governs flow of lubricant in the clearance space of a journal bearing is modified to study the laminar, transition and turbulent flows, by including turbulence coefficients $\overline{K_x}$ and $\overline{K_y}$. The SBA has been used to obtain the close form expression for pressure. This can be achieved by integrating twice the Reynolds equation and by applying the appropriate boundary conditions. Positive pressure zone was established by deleting sub ambient pressure. For super laminar flow, an iterative solution technique is used for establishing flow regimes; laminar, transition and fully developed turbulent flow. Methods for establishing equilibrium position of journal centre and for determining the static and dynamic performance characteristics, a FORTRAN program was developed.

4. Results and Discussion

The analysis and mathematical modeling was presented in the previous section, were used to compute the static and dynamic performance characteristics. The current study is based on L/D=0.25, and C/R=0.001 and on the assumption that the bearing and journal axes are parallel to each other. FORTRAN 77 compiler is used to developed the code based on the mathematical modeling discussed in section 2.1. The outputs obtained for various static and dynamic parameters from FORTRAN 77 compiler have been drawn graphically as show below.

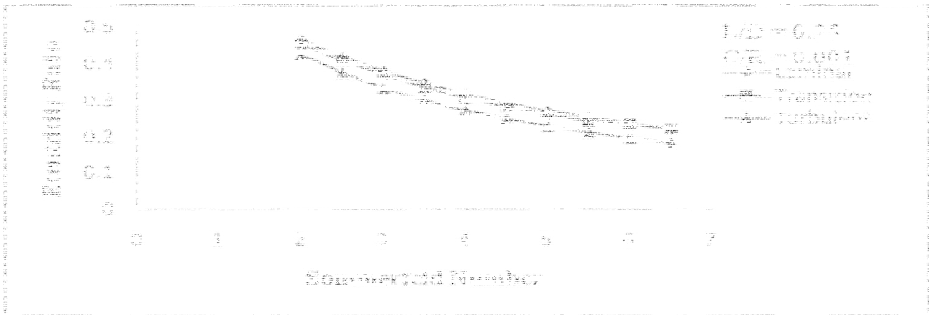

Fig.2 Eccentricity vs Sommerfeld number

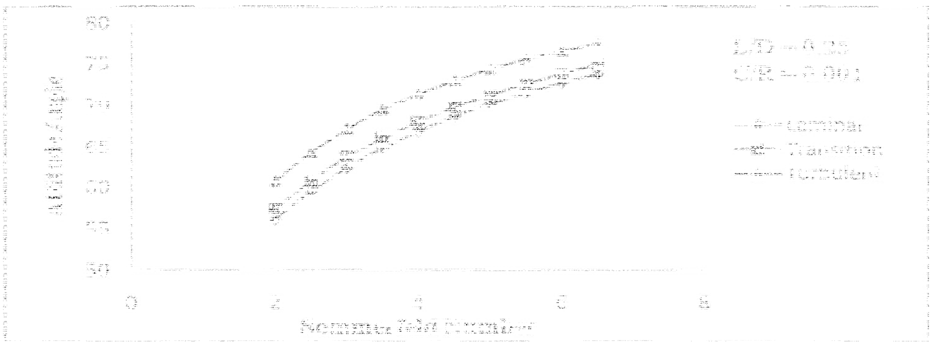

Fig.3 Attitude angle vs Sommerfeld number

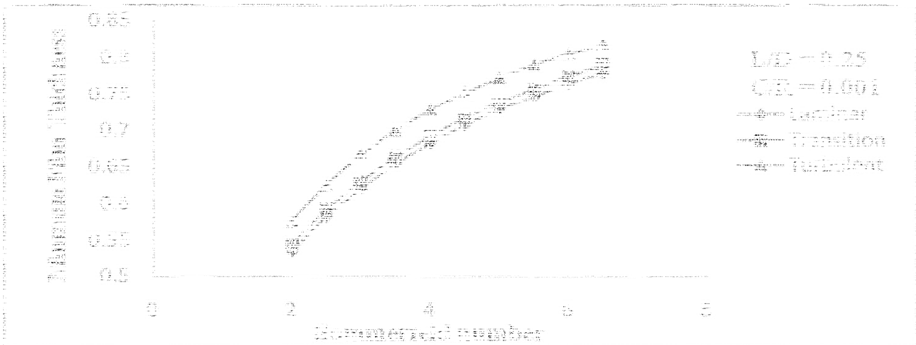

Fig.4 Minimum film thickness vs Sommerfeld number

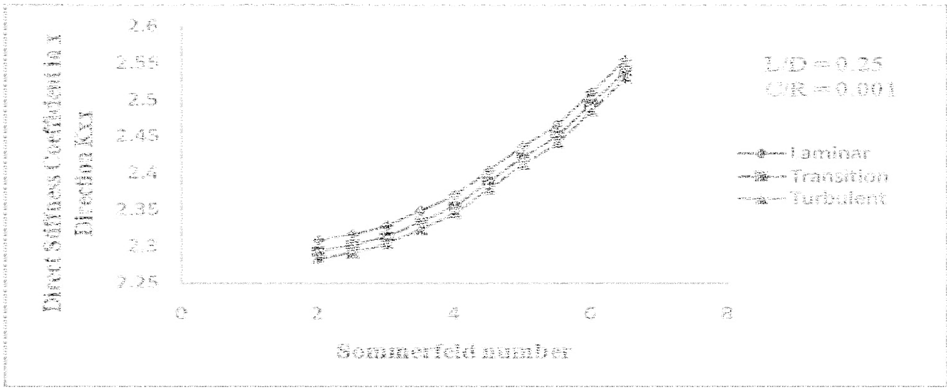

Fig.5 Dimensionless direct stiffness coefficient (K_{xx}) vs Sommerfeld number

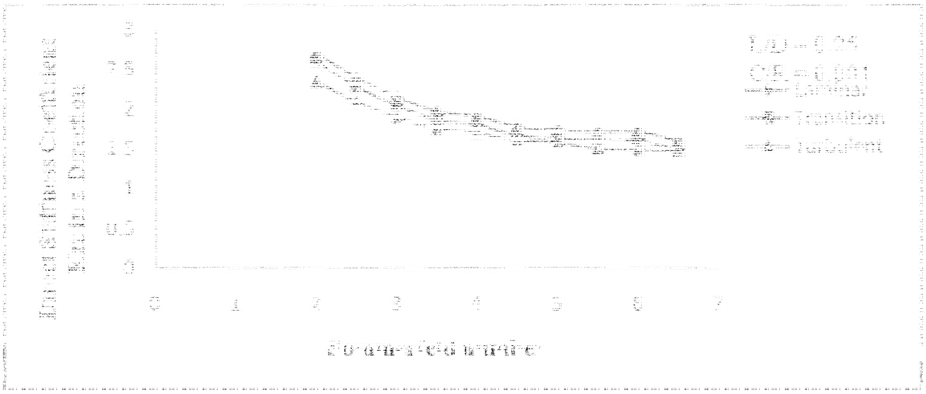

Fig.6 Dimensionless direct stiffness coefficient (K_{zz}) vs Sommerfeld number

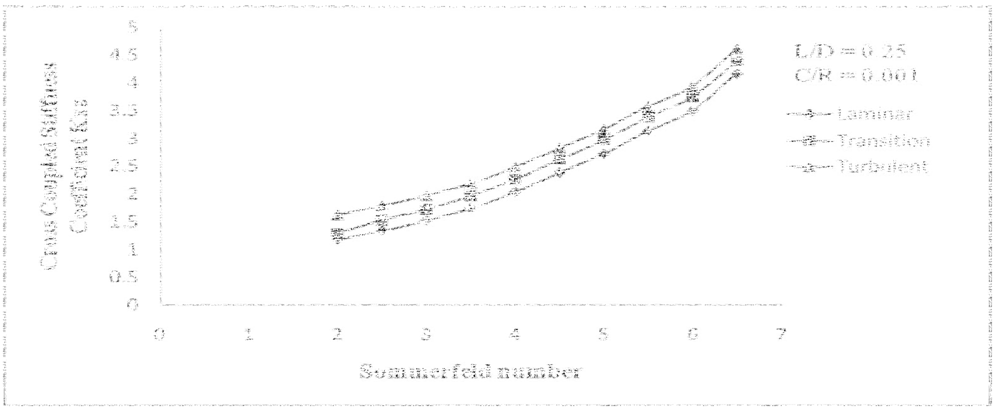

Fig.7 Dimensionless cross coupled stiffness coefficient (K_{zz}) vs Sommerfeld number

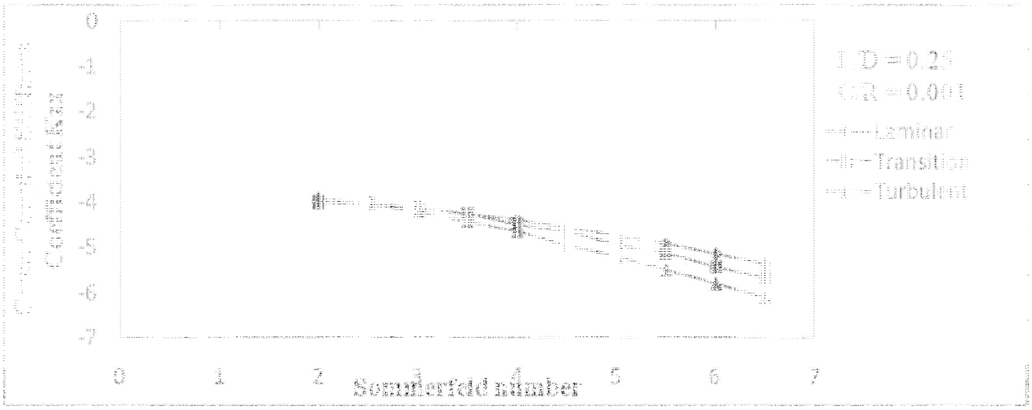

Fig.8 Dimensionless cross-coupled stiffness coefficient (K_{zx}) vs Sommerfeld number

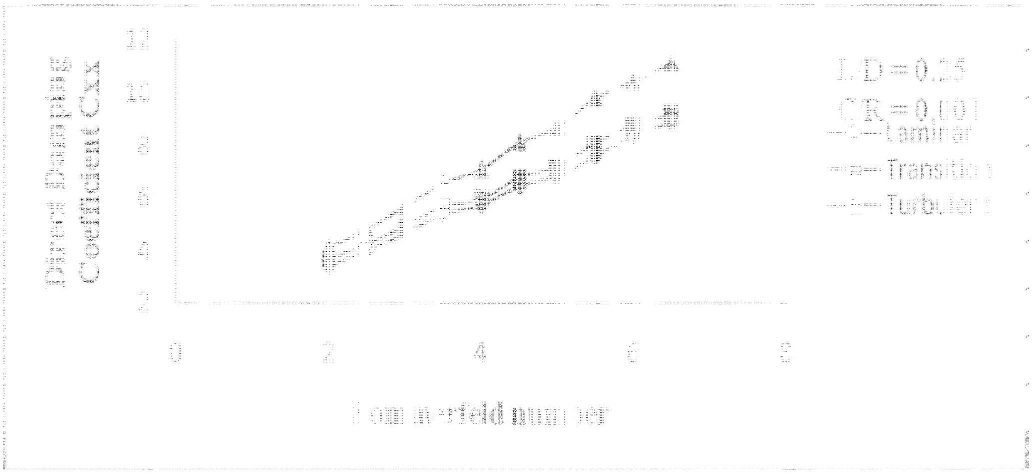

Fig.9 Dimensionless direct damping coefficient (C_{xx}) vs Sommerfeld number

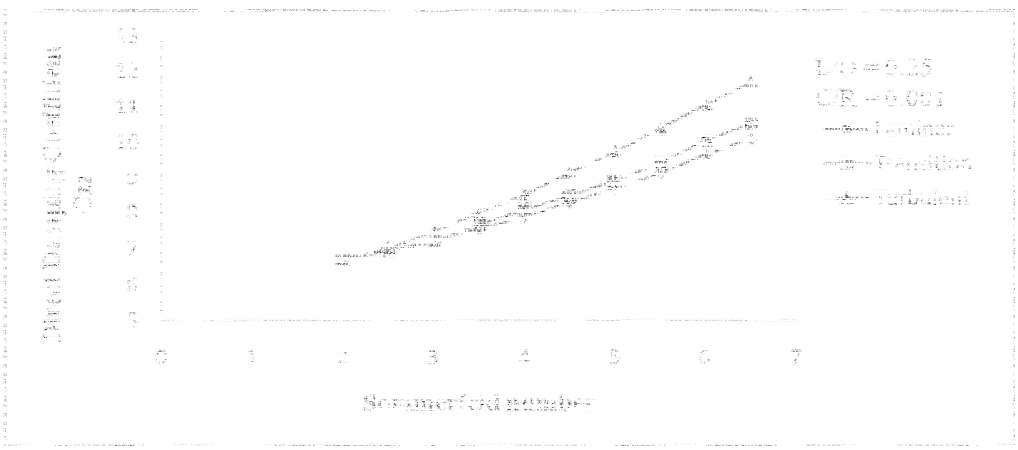

Fig.10 Dimensionless direct damping coefficient (C_{zz}) vs Sommerfeld number

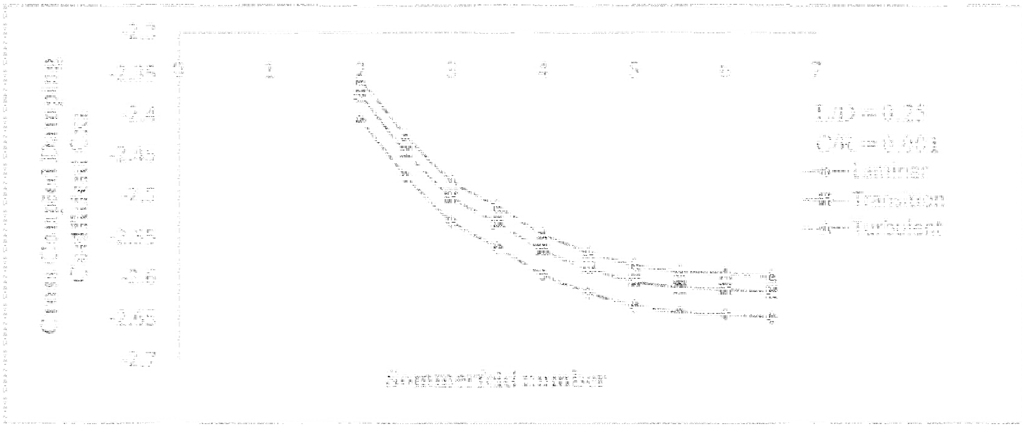

Fig.11 Dimensionless cross-coupled damping coefficient (C_{xz}= C_{zx}) vs Sommerfeld number

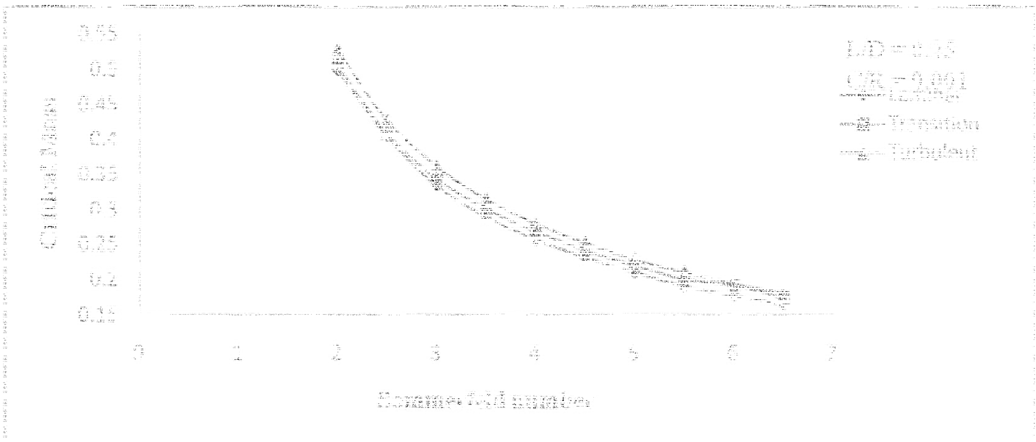

Fig.12 Critical Mass vs Sommerfeld number

There are several factors that influence the parameters of bearings like eccentricity ratio, attitude angle, minimum film thickness, spring and damping coefficients etc. with L/D and C/R ratios like type of lubricant flow. The variation of these parameters with Sommerfeld number have been drawn in above figures and discussed as follows.

1. Eccentricity ratio decreases with increase in Sommerfeld number for laminar, transition and turbulent flow regime of hydrodynamic journal bearing and at a constant Sommerfeld number and eccentricity ratio decreases while going from laminar to transition to turbulent flow as indicated in Fig 2.
2. Figure 3 reflects that the attitude angle increases with increases in Sommerfeld number for laminar, transition and turbulent flow regime of hydrodynamic journal bearing and at a constant Sommerfeld number attitude angle increases while going from laminar to transition to turbulent flow.
3. Minimum film thickness increases with increases in Sommerfeld number for laminar, transition and turbulent flow regime of hydrodynamic journal bearing and at a constant Sommerfeld number minimum film thickness increases while going from laminar to turbulent flow is reviled in Fig. 4.
4. Figure 5 and Fig.6 directs that stiffness coefficient Kxx increases while Kzz decreases with increase in Sommerfeld number for laminar, transition and turbulent flow regime of hydrodynamic journal bearing and at constant Sommerfeld number Kxx and Kzz decreases while going from laminar to transition to turbulent flow.
5. Cross coupled stiffness coefficient Kxz increases while Kzx decreases with increase in Sommerfeld number for laminar, transition and turbulent flow regime of hydrodynamic journal bearing and at constant Sommerfeld number Kxz increases and Kzx decreases while going from laminar from transition to turbulent flow is shown through Fig. 7 and Fig.8.

6. Figure 9 and Fig.10 shows direct damping coefficient Cxx and Czz increases with increase in Sommerfeld number for laminar, transition and turbulent flow regime of hydrodynamic journal bearing and at constant Sommerfeld number Cxx and Czz increases while going from laminar to transition to turbulent flow.
7. Cross coupled damping coefficient Cxz decreases with increase in Sommerfeld number in fluctuating manner for laminar, transition and turbulent flow regime of hydrodynamic journal bearing and at constant Sommerfeld number cross coupled damping coefficient increases while going from laminar to transition to turbulent flow is shown through Fig. 11.
8. Critical Mass decreases with increase in Sommerfeld number for laminar, transition and turbulent flow regime of hydrodynamic journal bearing and at constant Sommerfeld number critical mass increases while going from laminar to transition to turbulent flow is reflected through Fig. 12.

5. Conclusion:

Effects of different flow regime for example laminar, transient and fully developed turbulent on the static and dynamic performance parameter of hydrodynamic journal bearings was discussed analytically. Based on results, following conclusions can be drawn for a plane circular hydrodynamic bearing operating in laminar, transient and fully developed turbulent regime discussed for a constant L/D ratio = 0.25 and C/R ratio 0.001. As the fluid flow from laminar to turbulent the minimum film thickness, attitude angle Kxz, direct and cross coupled damping coefficients (Cxx, Czz, Cxz) increases for a constant Sommerfeld number. However, eccentricity ratio, Kxx, Kzx and Kzz decreases as the fluid moves in laminar to turbulent flow.

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